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COLD-AIR EXPERIMENTAL INVESTIGATION OF A TURBINE WITH BLADE TRAILING-EDGE COOLANT EJECTION

I - Single-Stage Turbine

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16. Abstract

Tests were made on a 0.762-meter- (30.0-in.-) tip-diameter research turbine to determine the effect of blade coolant flow on its aerodynamic performance. Both stator and rotor blades had trailing-edge slots for coolant ejection. Cooling air was supplied at the same pressure and nearly the same temperature as the turbine inlet air. The turbine was tested over a range of speed and pressure ratio. High primary efficiencies, calculated on the basis of primary air only, were obtained. At design speed and a pressure ratio of 1.755, the primary efficiency was 0.958. Attendant coolant fractions for the stator and rotor were 0.0524 and 0.0658, respectively. This efficiency was identical to that reported for the turbine from a previous investigation wherein only slotted (cooled) stator blades were incorporated in the turbine and tested. And it also compares with the 0.923 reported for the turbine with solid (uncooled) blading. Independently varying the rotor coolant flow, with other turbine operating conditions maintained as before, showed that rotor cooling imposed a severe penalty on turbine efficiency. A rotor coolant fraction above 0.063 was required before a net improvement in torque output (and primary efficiency) resulted. The thermodynamic efficiency, which accounts for the ideal energies of both blade coolant flows, decreased linearly with rotor coolant at a rate of about 0.7 percent per percent rotor coolant fraction.

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I - SINGLE-STAGE TURBINE

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SUMMARY

A cold-air investigation was conducted on the first stage of a two-stage, 0.762-meter- (30.0-in.-) tip-diameter turbine to determine the effect on its aerodynamic performance of cooling-air ejection from slots in the trailing edges of both stator and rotor blades. Coolant air was supplied to both blade rows at inlet pressures equal to the turbine inlet pressure. Tests were made over a range of speed and pressure ratio, and the performance results are presented herein. Additional tests were made at the design speed and a pressure ratio of 1.755. In these tests the rotor coolant flow was varied from zero to 0.08 that of the primary flow, while the stator coolant flow was regulated as before. Thus, the independent effect of rotor coolant flow is presented. These results are compared with those previously reported for the same basic turbine but with solid (uncooled) blading and also with those for the turbine with only slotted (cooled) stator blades. All tests were run at, or near, a primary- to coolant-air temperature ratio of unity.

It was found that, with coolant air supplied to both blade rows at a pressure equal to the turbine inlet pressure, the turbine yielded high primary-air efficiencies (defined herein) over the entire range of speed and pressure ratios investigated. A peak efficiency of over 0.96 was observed at overspeed conditions and in the very low pressure-ratio regime. At the design speed and a pressure ratio of 1.755, the turbine operating point at which the performances of the three differently cooled turbines are compared, the primary efficiency was 0.958. Attendant coolant fractions for the stator and rotor were 0.0524 and 0.0658, respectively. This efficiency was identical to that determined for the turbine wherein only stator cooling air was provided. And, it compares with the 0.923 for the uncooled (solid bladed) turbine.

The tests wherein the rotor coolant was varied showed that the addition of rotor coolant imposed a severe penalty on turbine efficiency. Thermodynamic efficiency decreased linearly with rotor coolant at a rate of about 0.7 percent per percent rotor coolant fraction. It required a rotor coolant fraction above 0.063 before a net improvement in torque output (and primary efficiency) was realized.

INTRODUCTION

Gas turbine engines used in advanced types of aircraft must have high turbine inlet temperatures in order to meet their mission objectives. These high temperatures usually require that the turbine blading be cooled. This, in turn, dictates that the turbine blades be thick, with blunt leading and trailing edges, in order to accommodate internal coolant passages. The resultant blades, then, represent a compromise from idealized aerodynamic design.

The NASA Lewis Research Center has been concerned with both the heat transfer and the aerodynamics of turbine-blade cooling methods. As part of this program, the aerodynamic effects on turbine performance of blade coolant flow discharge into the main gas stream is being experimentally investigated. The test turbine was modeled after a two-stage turbine for a high-temperature-engine application. The cold-air model had a 66.04-centimeter (26-in.) mean diameter with a 10.16-centimeter (4-in.) blade height in the first stage. Cold-air tests were made on a solid-bladed (uncooled) version of this turbine over a range of equivalent speed and pressure ratio. Similarly, the first stage of this research turbine was tested as a separate component. The results of these uncooled turbine tests are reported in references 1 and 2, respectively, and are used as a basis for comparison for subsequent cooled-turbine-blading configurations. To date, only the first stage of this two-stage turbine has been equipped with cooled stator blades and tested. The solid stator blading was replaced successively with three distinct cooled blade rows, all having the same blade aerodynamic profile but differing in the methods of coolant ejection. The first cooled stator configuration ejected coolant into the main gas stream through slots in the stator-blade trailing edges. The other two cooled stators were comprised of film-cooled blades: one set having discrete coolant holes, the other being fabricated with wire mesh supported internally. The same rotor was used for all three test programs. A brief description of the bladings tested and a summary of the comparative single-stage turbine performance results are given in reference 3. Therein it was reported that the turbine with slotted stator-blade trailing edges obtained the highest stage efficiency at comparable turbine operating conditions. Further, this configuration resulted in more coolant flow than did the two transpiration-type stator bladings. With these considerations, the solid-bladed rotor was replaced with one incorporating hollow blades of the same profile as before and also having trailing-edge slots. This report, then, presents results of tests on this single-stage turbine with coolant-flow ejection from both stator- and rotor-blade trailing edges.

The test program was divided into two phases. First, the turbine was operated over a range of equivalent speed and pressure ratio with coolant flow supplied to both blade rows at a pressure equal to the turbine inlet pressure (10.16 N/cm^2; 30.0 in. Hg abs). Both the primary and coolant air were supplied by the laboratory combustion air system. The primary air was heated and maintained at a turbine inlet temperature of 378 K

(680° R). The coolant air was unheated and varied in temperature from 286 to 305 K (514° to 549° R). Performance parameters are presented in terms of mass flow, torque, and efficiency. Coolant fractions (ratio of coolant flow to primary flow) for both blade rows are also presented.

The second phase of testing was conducted to determine the effect on turbine performance of varying only the rotor coolant flow, with stator coolant flow maintained as for the phase 1 tests. These tests were made at equivalent design speed, at a turbine pressure ratio of 1.755, and over a range of rotor coolant fractions from zero to 0.08 of primary flow.

Test results are compared with those obtained for the base (uncooled) turbine (ref. 2) and with those for the turbine equipped with the slotted stator blades. A more comprehensive discussion of results from the latter test program is presented in reference 4. These reference tests were conducted with nominally ambient turbine inlet temperature. And, the stator coolant inlet temperature, as used in reference 4, was also ambient. Although the subject tests were run at a primary- to coolant-air temperature ratio of about 1.3, this value should be sufficiently close to the reference 4 value of unity such that heat transfer effects would be negligible, and hence, are ignored herein.

The work was performed in the U.S. customary system of units. Conversion to the International System of Units was for reporting purposes only.

SYMBOLS

A annular flow area, m^2; ft^2

g force-mass conversion constant, unity in SI system; 32.174 ft/sec^2

h specific enthalpy, J/kg; Btu/lbm

J mechanical equivalent of heat, unity in SI system; 778.16 ft-lbf/Btu

N rotational speed, rpm

p absolute pressure, N/m^2; lbf/ft^2

R gas constant: for mixture of air and combustion products used herein, 288 J/(kg)(K), or 53.527 ft-lbf/(lbm)(°R); for cooling air, 287 J/(kg)(K), or 53.342 ft-lbf/(lbm)(°R)

T temperature, K; °R

U blade velocity, m/sec; ft/sec

V absolute gas velocity, m/sec; ft/sec

W gas velocity relative to rotor blade, m/sec; ft/sec

w mass flow rate, kg/sec; lbm/sec

y coolant fraction, ratio of coolant flow to primary flow

α absolute flow angle (measured from axial), positive in direction of rotor rotation, deg

γ ratio of specific heats: 1.398 for mixture of air and combustion products used herein; 1.400 for cooling air

δ ratio of turbine inlet total pressure to U.S. standard sea-level pressure of 1459 N/cm^2 (2116.22 lb/ft^2 abs)

ϵ function of γ, $\dfrac{0.73959}{\gamma}\left[\left(\dfrac{\gamma+1}{2}\right)^{\gamma/(\gamma-1)}\right]$

η efficiency based on total-pressure ratio

θ_{cr} ratio of critical velocity at turbine inlet to critical velocity (V_{cr} = 310.62 m/sec (1019.1 ft/sec)) at U.S. standard sea-level air temperature of 288.17 K (518.7° R)

τ torque, N-m; ft-lbf

Subscripts:

c coolant flow

cr conditions at Mach 1 (critical)

h hub radius

id ideal

m mean blade height

o zero coolant flow

p primary flow

r rotor

s stator

t tip radius

th thermodynamic

u uncooled

x axial component

0 measuring station at turbine inlet (fig. 5)

1 measuring station at stator outlet

2 measuring station at rotor outlet

Superscript:

' total state

TURBINE DESIGN

The design requirements and physical features of the first stage of the two-stage research turbine are described in detail in reference 5. This stage is typical of the first stage of a turbine for an advanced high-temperature engine. Some equivalent design requirements are restated herein for the convenience of the reader as follows:

Equivalent specific-work output, $\Delta h/\theta_{cr}$, J/g; Btu/lbm 39.57; 17.00
Equivalent mean blade speed, $U_m/\sqrt{\theta_{cr}}$, m/sec; ft/sec. 152.4; 500.0
Design equivalent mass flow, $w_p\sqrt{\theta_{cr}}/\delta$, kg/sec; lbm/sec 18.1; 39.9

From these the following operating parameters are derived:

Equivalent design speed, $N/\sqrt{\theta_{cr}}$, rpm. 4407.36
Equivalent design torque, $\epsilon\tau/\delta$, N-m; ft-lbf 1550.5; 1143.6
Equivalent design mass-flow - speed parameter, $\epsilon w_p N/\delta$,
(kg)(rad)/sec^2; (lbm)(rpm)/sec 8352.0; 175 854

A 0.762-meter- (30.0-in.-) tip-diameter turbine was selected; blade heights were 10.16 cm (4.0 in.). The first-stage velocity diagrams evolved to meet these design requirements are shown in figure 1. The velocity-diagram vector velocity values are included. These values represent the free-stream uniform flow conditions entering and leaving the rotor.

The blade-surface velocity distributions for the stator and rotor blading are reproduced from reference 5 and shown in figure 2. Stator- and rotor-blade coordinates are given in table I.

APPARATUS AND INSTRUMENTATION

The "design" stator blades were fabricated hollow. The blade trailing edges were modified as shown in figure 3(a) for the investigation of reference 4. The rounded trailing edges of the stator blades were milled square, and slots were machined to the hollow cores. This procedure is described in detail in reference 6. A cutaway view of a modified stator blade is shown in figure 3(b). A closeup of the slotted-stator-blade assembly

installed in the test facility is shown in figure 4. The turbine was tested with coolant flow ejection from these slotted stator blades; the results are presented in reference 4. The same stator was used for the subject investigation.

The rotor and rotor blading of the turbine investigated and reported in reference 4 were replaced with an assembly to provide for cooling air. New hollow blades were fabricated to the same blade profile and capped, and the trailing edges were slotted. A sketch of a cooled rotor blade is shown in figure 5; the slotted-rotor-blade assembly is shown in figure 6. The stator had seven structural struts in the trailing-edge section, as compared to three for the rotor. These struts can be noted in figures 4 and 6. Since both the stator- and rotor-blade heights were about the same (except for the 0.076-cm (0.030-in.) rotor-tip clearance), each trailing-edge coolant slot area was basically the same (within 1 percent).

The turbine test facility is shown in figure 7. This facility was basically the same as that used for tests of the turbine with the slotted stator blades (ref. 4) but was modified to provide for the added rotor cooling air. Further, the primary-air heater used in the two-stage turbine investigation (ref. 1) was also used in the subject turbine tests. The rotor coolant flow was supplied in the same manner as was the stator coolant flow. That is, it also was supplied from the laboratory combustion air system and passed through a venturi flowmeter with associated instrumentation. A downstream throttle valve was used to regulate coolant flow. The flow was then piped through the tailcone to the downstream face of the rotor, through a carbon-steel face seal, into a hollow rotor, through a hole in the blade base (fig. 5), into the blade cavity, and was discharged through the trailing-edge slots into the main gas stream. A cross-sectional view of the research turbine, showing the two blade cooling systems, is presented in figure 8.

The research instrumentation was essentially the same as reported in reference 4 with the additional aforementioned rotor coolant mass-flow-rate venturi. The state of the stator coolant inlet air was measured in the supply annulus immediately over the stator blades (fig. 8); the state of the rotor coolant inlet air was determined from pressure and temperature measurements taken in the horizontal run of the supply pipe inside the tailcone and along the centerline of the turbine. Other instrumentation measured primary mass flow, total pressures, static pressures and temperatures at the turbine inlet, static pressures at the stator exit, static pressures and flow angles at the turbine exit, rotative speed, and output torque. The research-instrumentation measuring stations are shown in figure 8. Which specific measurements of pressure, temperature, and flow angle were taken at each of these measuring stations are indicated in the instrumentation plan of figure 9.

All research instrumentation was connected to a 100-channel data acquisition system which measured and recorded on paper tape the electrical signals from the respective transducers. For each steady-state turbine operating point, a minimum number of five

readings from each transducer were recorded. These readings were subsequently numerically averaged.

PROCEDURE

The test program on the research, single-stage, cooled turbine was conducted in two phases. In phase 1, performance data were taken over a range of overall pressure ratio and speed with coolant air supplied to both stator and rotor blades at a pressure equal to the turbine inlet pressure (10.16 N/cm^2; 30.0 in. Hg abs). In the phase 2 tests, the rotor coolant flow was independently varied, and other turbine test conditions were maintained the same. Both tests were conducted with the turbine inlet temperature maintained constant at 378 K (680° R). At this temperature, the operating test speed corresponding to the equivalent design speed was 5053 rpm.

In test phase 1, the turbine speed was varied from 40 to 110 percent of design speed in 10-percent increments. The total-pressure ratio was varied from about 1.4 to 2.4. Pressure-ratio changes were made by adjusting the turbine outlet pressure through regulation of valves in the laboratory altitude exhaust system. The cooling-air temperature varied from 286 to 292 K (514° to 525° R).

Phase 2 tests were conducted at the design speed and a pressure ratio of 1.755. It was at this pressure ratio that the turbine with the slotted stator blades and the solid rotor blades developed the stage design equivalent specific-work output of 39.57 joules per gram (17.00 Btu/lbm) with zero stator coolant flow. Corresponding test results are presented in reference 4, in which the effect on turbine performance of varying only stator coolant flow is reported. The subject tests, then, were made to determine the added effect of varying the rotor coolant flow. In these tests, both the stator and rotor coolant flows were supplied at 304±1 K (546° to 549° R), somewhat higher than temperatures encountered in the phase 1 tests.

Turbine performance was based on total-pressure ratio. Turbine inlet total pressure p_0' was calculated from static pressure, primary-air mass flow, the known annulus area, and the total temperature by the following equation:

$$p_0' = p_0 \left[\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma}\left(\frac{w_p}{p_0 A_0}\right)^2 RT_0'}\right]^{\gamma/(\gamma-1)} \tag{1}$$

Turbine outlet total pressure p_2' was similarly calculated by using static pressure, turbine exit flow angle, annulus area, and total temperature but included the sum of the primary and coolant flow (or flows), such that

$$p_2' = p_2 \left[\frac{1}{2} + \sqrt{\frac{1}{4} + \frac{\gamma - 1}{2g\gamma} \left(\frac{w_p + w_c}{p_2 A_2} \right)^2 \frac{RT_2'}{\cos^2 \alpha_2}} \right]^{\gamma/(\gamma-1)} \tag{2}$$

The turbine outlet total temperature T_2' was derived from the inlet temperature, torque, mass flow, and speed. The outlet flow angle α_2 used in equation (2) is the average divergence from the axial direction, irrespective of sign.

Two efficiencies are defined for use in this report: primary efficiency η_p and thermodynamic efficiency η_{th}. In equation form,

$$\eta_p = \frac{2\pi \tau N/60J}{w_p \, \Delta h_{id,p}} \tag{3}$$

$$\eta_{th} = \frac{2\pi \tau N/60J}{w_p \, \Delta h_{id,p} + w_{c,s} \, \Delta h_{id,c,s} + w_{c,r} \, \Delta h_{id,c,r}} \tag{4}$$

Primary efficiency relates the total power output of the primary and coolant flow (or flows) to the ideal power of only the primary flow. The thermodynamic efficiency takes into account the ideal energy of the coolant flow (or flows).

RESULTS AND DISCUSSION

The results of the experimental investigation are discussed in three parts. First, the overall performance results of the turbine when it is operated over a range of speed and pressure ratio with coolant air supplied to both stator and rotor blading at pressures equal to the turbine inlet pressure are discussed. Second, test results obtained at the design speed and a pressure ratio of 1.755 with the stator coolant supplied as in phase 1 but with the rotor coolant flow varied from zero to 0.08 of the primary flow are discussed. All data and test results are shown in terms of equivalent air values. Results are then compared with results from investigations of the uncooled (solid bladed) turbine (ref. 2) and the turbine with only stator cooling (ref. 4). The cooled-turbine tests were conducted with a primary- to coolant-air temperature ratio at, or slightly above, unity.

Overall Turbine Performance

The overall turbine performance is presented in terms of primary mass flow, blade coolant mass flows, torque output, and the resultant performance map. Additional data

nclude the flow angle at the turbine exit as a function of turbine rotative speed and over-ll total-pressure ratio. Efficiency data for design speed and for a range of pressure atio are shown. The static-pressure distribution at the three measuring stations rough the turbine is also presented over the same range of test variables.

Mass-flow characteristics. - The variation of equivalent primary mass flow $w_p \sqrt{\theta_{cr}}/\delta$ with overall total-pressure ratio p_0'/p_2' for the equivalent speeds $N/\sqrt{\theta_{cr}}$ nvestigated is shown in figure 10. Primary flow increased with pressure ratio for all peeds, until choking (constant) values were reached for the intermediate and high peeds. In the choked pressure-ratio regime the flow decreased with increasing speed, ndicating that the rotor, not the stator, limited the flow.

These turbine tests were conducted with both the stator and rotor coolant flows sup-lied at turbine inlet pressure (10.16 N/cm^2; 30.0 in. Hg abs). The resultant stator oolant fraction $w_{c,s}/w_p$ and rotor coolant fraction $w_{c,r}/w_p$ are shown in fig-res 11(a) and (b), respectively, as functions of total-pressure ratio and equivalent peed. These data indicate that with the same coolant supply pressures, a higher cool-nt flow results for the rotor than for the stator. In figure 11(a), for a given pressure atio a slight decrease in stator coolant fraction with increasing rotor speed prevails. onversely, figure 11(b) shows a relatively larger increase in rotor coolant fraction with ncreasing rotor speed. The total coolant fraction over the entire range of speed and ressure ratio tested varied only from 0.11 to 0.12. The larger rotor coolant fraction esults primarily from the fact that there were 61 slotted rotor blades as compared ith 50 slotted stator blades.

Torque characteristics. - Figure 12 presents the variation of equivalent torque out-ut $\epsilon\tau/\delta$ with pressure ratio for the equivalent speeds investigated. The data are typ-cal, the torque increasing with pressure ratio for all speeds. Limiting blade loading, efined as that point where increasing the pressure ratio results in no increase in torque utput, did not occur at any speed.

Performance map. - Data from figures 10 and 12 were used to evolve the cooled-urbine performance map shown in figure 13. Equivalent primary specific-work output h/θ_{cr} is shown as a function of a primary-mass-flow - speed parameter $\epsilon w_p N/\delta$ for nes of constant total-pressure ratio and equivalent rotor speed. Contours of constant alues of primary efficiency η_p (eq. (3)), based on the total-pressure ratio across the urbine, are also superimposed. Turbine efficiency is seen (fig. 13) to range from about .80 in the low-speed regime to above 0.96 at 110-percent speed. The entire level of fficiency was high (1) because the stator cooling air did indeed add to the turbine work utput in passing through the rotor (ref. 4) and (2) because the efficiency was based only n the primary mass flow (eq. (3)). At the equivalent design speed the primary effi-iency varied from about 0.92 to 0.958. The solid symbol denoting a pressure ratio of .755 corresponds to that operating point where the single-stage turbine with the slotted

stator blades and zero coolant flow along with the solid (uncooled) rotor blades obtained the stage design equivalent specific-work output of 39.57 joules per gram (17.00 Btu/lbm) at design speed (ref. 4). For the subject turbine, the equivalent specific-work output at this pressure ratio was 41.23 joules per gram (17.71 Btu/lbm); the attendant efficiency was 0.958. The corresponding equivalent primary mass flow was 18.60 kilograms per second (41.01 lbm/sec), as obtained from figure 10. The stator and rotor coolant fractions were 0.0524 and 0.0658, respectively (fig. 11). The performance of this turbine will subsequently be compared with the performances of the uncooled turbine (ref. 2) and the turbine with only stator cooling (ref. 4).

Efficiencies at equivalent design speed. - Both primary and thermodynamic efficiencies, as defined herein, are presented in figure 14 as a function of turbine overall total-pressure ratio for the equivalent design speed. The thermodynamic efficiencies were lower by about 0.08 over the range of pressure ratios tested. This, of course, reflects the inclusion of the ideal work of both coolant flows in the thermodynamic efficiency equation (eq. (4)), as differentiated from the primary efficiency (eq. (3)). Both efficiencies increased slightly with pressure ratio and peaked at a pressure ratio of about 1.8, followed by a more marked decrease with further increases in pressure ratio. At the reference pressure ratio of 1.755, indicated on the abscissa of figure 14, the primary-air efficiency of the cooled turbine was about at its peak value of 0.958 (as noted on the performance map (fig. 13)); the corresponding thermodynamic efficiency was 0.878.

Outlet flow angle. - The variation of turbine outlet flow angle α_2 is shown in figure 15 as a function of the equivalent speed and the total-pressure ratio across the turbine. Negative angles correspond to a positive contribution to turbine work output. The spacing between the low-speed curves is inconsistent with similar results obtained in the uncooled-turbine investigation of reference 2. When the turbine with only the slotted stator was tested (ref. 4), only slight inconsistencies in the angle measurements were observed but were not noted in the reference. Now, all these outlet flow angle measurements were made $1\frac{1}{2}$ axial rotor-blade chord lengths downstream of the rotor-blade trailing edges. Apparently, this distance was insufficient, and the observed angle measurements were influenced by the trailing-edge-ejected coolant flows, such that true free-stream aftermixed flow angles were not measured.

The outlet flow angle measurements were reflected in the calculated outlet total pressure p_2' (eq. (2)) and hence the total-pressure ratio. Further, at a given pressure ratio, the absolute value of outlet flow angle (fig. 12) significantly increased with decreasing speed. Hence, in the low-speed regime, where turbine exit flow angles are large, the cosine term in equation (2) is more sensitive to small angle changes, thereby affecting the pressure ratio. The outlet flow angle data for the higher speeds (fig. 15) appear to be consistent. At the equivalent design speed and the pressure ratio of 1.755, the outlet flow angle was -14.6°. This angle was sufficiently close to axial such that the

calculated outlet total pressure (hence, pressure ratio and efficiency) was relatively insensitive to it.

That pressure ratio was affected by outlet flow angle for the low-speed tests can also be noted on the performance map (fig. 13), where inflections in the pressure-ratio lines resulted. The curvature of these pressure-ratio lines, in turn, affects the efficiency contours. The trends are indicated. In view of the preceding discussion, and particularly for turbines with cooled turbine blading, care should be exercised (1) when locating axially the angle probes in the turbine outlet section and (2) when turbine results are being interpreted.

Static-pressure distribution. - The variation in static pressure at the three measuring stations through the turbine is shown in figure 16 as a function of total-pressure ratio for equivalent design speed. The static-pressure measurements at the hub are presented in figure 16(a); the tip measurements in figure 16(b). All data were normalized to the inlet total pressure. Choking in a rotor blade row is indicated when the static pressure at the inlet to the blade row remains constant while the static pressure at the exit of the blade row continually decreases with increasing pressure ratio. As shown in figure 16 the hub and tip sections of the rotor choked at a total-pressure ratio of about 2.1. This is in agreement with the design-speed, mass-flow data presented in figure 10, which also showed the rotor to choke at the same pressure ratio. This choking pressure ratio is well above the pressure ratio of 1.755 used herein for comparison purposes. Static-pressure data for other speeds exhibited similar characteristics and are not presented.

Effect of Variable Rotor Coolant Flow

The test to determine the effect of rotor coolant flow was conducted at the equivalent design speed and a pressure ratio of 1.755. As stated previously, it was this pressure ratio at which the turbine with the slotted stator blades and the uncooled (solid) rotor blades obtained the equivalent design specific-work output of 39.57 joules per gram (17.00 Btu/lbm) with zero stator coolant flow (ref. 4). Throughout these tests, cooling air was supplied to the stator blades at a pressure equal to the turbine inlet total pressure (10.16 N/cm^2; 30.0 in. Hg abs). The coolant flow to the rotor blades was then varied from zero to about 0.08 of the primary flow. The cooling air was supplied by the laboratory combustion-air system at a temperature of about 304 K (548° R) for this test, considerably higher than the temperatures for the preceding tests. Turbine inlet temperature was again maintained at 378 K (680° R).

Efficiency with zero rotor coolant. - In order to show the effect on turbine efficiency of adding rotor coolant, it is necessary to first establish the efficiency with no rotor coolant flow. However, tests run under this condition resulted in a performance

degradation because of the presence of the rotor trailing-edge slots with probable internal recirculation of main-stream and boundary-layer air. An assumption was therefore made that the efficiency of the stator coolant, with appropriate corrections for turbine inlet temperature differences, was the same for the subject turbine as for the turbine of reference 4, wherein similar tests were made with the same configuration, the slotted stator along with the solid rotor. Primary efficiency (eq. (3)) is rewritten as

$$\eta_p = \frac{\Delta h_p}{\Delta h_{id,p}} + \left(\frac{w_c}{w_p}\right)\left(\frac{\Delta h_c}{\Delta h_{id,c}}\right)\left(\frac{\Delta h_{id,c}}{\Delta h_{id,p}}\right)$$

or

$$\eta_p = \eta_{pa} + y\eta_{ca}\left(\frac{T_c'}{T_{0,p}'}\right)\left[\frac{1 - \left(p_2'/p_c'\right)^{(\gamma-1)/\gamma}}{1 - \left(p_2'/p_{0,p}'\right)^{(\gamma-1)/\gamma}}\right] \tag{5}$$

where

η_{pa} efficiency of primary air

η_{ca} efficiency of coolant air

Now, at an overall pressure ratio of 1.755, a stator coolant- to primary-air inlet pressure ratio of 1.0, and a primary- to coolant-air inlet temperature ratio of 1.0, the measured primary efficiency from reference 4 was 0.958 and the stator coolant fraction was 0.0468. With zero coolant flow the measured efficiency of the primary air η_{pa} was 0.920. The resulting efficiency of the cooling air η_{ca} for these reference conditions and from equation (5) was 0.812.

For the subject turbine, the primary air was heated to 378 K (680° R), as compared to 304 K (548° R) for the coolant inlet air. At a primary- to coolant-air inlet pressure ratio of unity, the stator coolant fraction increased, relative to that of reference 4, from 0.0468 to 0.0512 because of the lower density of the primary air. With a constant efficiency of 0.812 for the stator coolant flow, the estimated primary efficiency (eq. (5)) for the subject turbine with zero rotor coolant η_o was 0.954. The accompanying thermodynamic efficiency was determined to be 0.916.

Efficiency with variable rotor coolant. - The fractional change in primary and thermodynamic efficiencies $(\eta - \eta_o)/\eta_o$ as a function of rotor coolant is shown in figure 17 for design speed and the reference pressure ratio of 1.755.

The thermodynamic efficiency decreased linearly with rotor coolant at a rate of about 0.7 percent per percent coolant. This was a large penalty which, of course,

resulted from the fact that the only source of energy from the coolant was jet-reaction work as it exited from the rotor-blade trailing edges.

The variation of primary efficiency is an indication of the torque energy contributed by the coolant. It is analogous to the variation in kinetic energy output of a vane (ref. 7). If primary efficiency increases, the rotor coolant has a net effect of adding torque. If it is the same as the efficiency at zero rotor coolant, the net effect is a zero change in torque. Finally, if primary efficiency decreases, not only does the coolant add no torque, but also it requires torque developed by the primary air to pump it through the disk and to eject it from the blade trailing edges to the main-stream conditions.

The primary efficiency curve of figure 17 shows that adding rotor coolant resulted in a reduction in net torque output below a coolant fraction of 0.063. Adding 6.3-percent coolant resulted in no contribution of torque from the coolant, which is a severe penalty for a single-stage turbine. Of course, some of the energy ejected from the rotor could be recovered in succeeding stages of a multistage turbine or by the low-pressure turbine in a turbofan engine. Above a rotor coolant fraction of 0.063, figure 17 shows that the rotor coolant contributed to torque output.

The physical significance of what happens as rotor coolant is added can be seen more clearly by isolating the individual energies involved in primary efficiency, which may be defined as

$$\eta_p = \frac{w_p \, \Delta h_p + w_{c,r} \, \Delta h_{jet} - w_{c,r} \, \Delta h_{pump}}{w_p \, \Delta h_{id,p}}$$

or

$$\eta_p = \eta_{pa} + y_r \frac{\Delta h_{jet}}{\Delta h_{id,p}} - y_r \frac{\Delta h_{pump}}{\Delta h_{id,p}} \qquad (6)$$

If it is assumed that the efficiency of the primary air η_{pa} remains constant and equal to that at zero rotor coolant fraction η_o, then

$$\frac{\eta_p - \eta_o}{\eta_o} = y_r \frac{\Delta h_{jet}}{\eta_o \, \Delta h_{id,p}} - y_r \frac{\Delta h_{pump}}{\eta_o \, \Delta h_{id,p}}$$

and

$$\frac{\Delta \eta_p}{\eta_o} = y_r \frac{\Delta h_{jet}}{\Delta h_p} - y_r \frac{\Delta h_{pump}}{\Delta h_p} \qquad (7)$$

The first term on the right side of equation (7) is equal to the change in primary efficiency due to the jet power of the ejected coolant, or

$$\left(\frac{\Delta\eta_p}{\eta_o}\right)_{jet} = y_r \frac{\Delta h_{jet}}{\Delta h_p}$$

and is approximated as

$$\left(\frac{\Delta\eta_p}{\eta_o}\right)_{jet} = y_r \frac{U(W_{jet} \sin \beta_2 - U)}{gJ\ \Delta h_p} \tag{8}$$

where W_{jet} is the coolant-air exit velocity relative to the rotor blade and β_2 is the coolant-air exit relative flow angle measured from the axial direction. Using the familiar expression for speed-work parameter ($\lambda = U^2/gJ\ \Delta h$), equation (8) can also be written as

$$\left(\frac{\Delta\eta}{\eta_o}\right)_{jet} = \lambda y_r \left[\left(\frac{W_{jet}}{U}\right) \sin \beta_2 - 1\right] \tag{8a}$$

As coolant is added, equation (8a) indicates that jet power can be negative or positive depending on the magnitude of W_{jet}. Pump power is always negative and is approximated by

$$\left(\frac{\Delta\eta}{\eta_o}\right)_{pump} = y_r \frac{U^2}{gJ\ \Delta h_p} \tag{9}$$

or

$$\left(\frac{\Delta\eta}{\eta_o}\right)_{pump} = \lambda y_r \tag{9a}$$

The effect of pump power (eq. (9a)) was calculated from data; the effect of jet power was determined by adding pump power to measured overall values of $\Delta\eta_p/\eta_o$ by equation (7).

The individual effects of pump and jet power on primary efficiency are shown in figure 18. Also shown in figure 18(b) are three rotor coolant jet flow velocity diagrams

superimposed on the primary-air velocity diagrams. At low values of W_{jet} (coolant fractions below 0.033), the exit absolute jet velocity V_{jet} was in the direction of rotor rotation and hence reduced torque output. At a coolant fraction of 0.033, V_{jet} was axial ($W_{jet} \sin \beta_2 = U$ in eq. (8)) and jet power was zero. This is also the minimum point on the primary efficiency curve of figure 17. For coolant fractions above 0.033, V_{jet} added to work output and consequently added to torque. At a coolant fraction of 0.063 (fig. 18), the positive contribution of jet power was equal to the negative contribution of pump power and, as mentioned earlier in the discussion of figure 17, the net output was the same as that for zero rotor coolant. For coolant fractions above 0.063, jet power exceeded pump power and resulted in a net increase in torque caused by the coolant.

In summary, the effect of rotor coolant on turbine performance can be very significant, especially for single-stage turbines. Although the ejected coolant may produce torque in downstream stages of multistage turbines, the energy required to pump the coolant up to wheel speed and the probable negative work contribution of the ejected coolant makes it highly desirable to minimize the amount of rotor coolant required. Efficient means of pumping the coolant up to blade speed, such as ejecting it through tangentially oriented nozzles near the disk rim, are being studied. However, all energies and losses involved must be accounted for by considering the overall effect of such schemes on the engine.

Comparison of Cooled- and Uncooled-Turbine Performance

Although the subject report concerns a single-stage turbine with coolant flow ejection from the trailing edges of both stator and rotor blades, some results were compared with results for the uncooled (solid bladed) turbine (ref. 2) and with results for the turbine wherein only the stator blades used cooling air (ref. 4). For completeness, as well as for the convenience of the reader, these results are included in table II, wherein the performance of the three turbine configurations are compared at the equivalent design speed and a turbine pressure ratio of 1.755 and for various stator and rotor coolant flow conditions where applicable.

For ease in the ensuing discussion, each turbine was assigned a number. The uncooled turbine is turbine 1, the turbine with stator-blade cooling is turbine 2, and the subject turbine is turbine 3. The alphabetical designations added to turbines 2 and 3 (table II) refer to different modes of cooling. For each case where coolant flow was used, the tabulated coolant fractions were obtained when the coolant supply pressure (or pressures) was equal to the turbine inlet pressure (10.16 N/cm^2; 30.0 in. Hg abs). Turbines 1 and 2 were tested with basically ambient-air turbine inlet temperature. Turbine 3 was tested with the inlet air heated to 378 K (680° R). There may appear to be a redundancy between turbines 3b and 3c of table II. However, these two test programs

were run at different times, and the blade coolant-air temperatures were different. Turbine performance results were sufficiently affected by this temperature difference as to warrant inclusion in table II and subsequent discussion.

In this section the primary mass-flow rates and primary efficiencies obtained for the three turbine configurations are compared at equivalent design speed and over the range of pressure ratios tested. Changes in blade-row reaction characteristics for the three turbines at equivalent design speed and a pressure ratio of 1.755 are also discussed.

Design-point comparison. - Table II shows pertinent performance results for turbine 1 (the solid bladed, or uncooled, turbine) from reference 2. The data shown correspond to the turbine operating point where equivalent design specific-work output (39.57 J/g; 17.00 Btu/lbm) was obtained at equivalent design speed. The attendant pressure ratio was 1.751, and the turbine efficiency was 0.923. Then, with the stator blades modified to include cooling-air slots in the trailing edges (turbine 2a), and at the same turbine work output, the efficiency decreased slightly to 0.920 and occurred at a pressure ratio of 1.755 (ref. 4). The stator modification had an insignificant effect on the equivalent primary flow and the outlet flow angle. The net effect of blade cooling air, then, can be determined if this pressure ratio (1.755) is used as the basis of comparison for the differently cooled turbine configurations.

It is reported in reference 4 that when cooling air was supplied to the slotted stator blades at a pressure equal to the turbine inlet pressure (turbine 2b, table II), the resultant stator coolant fraction was 0.0468 at the turbine pressure ratio of 1.755. With this stator coolant flow, the turbine equivalent specific-work output was increased from 39.57 (turbine 2a, table II) to 41.30 joules per gram (17.00 to 17.74 Btu/lbm). The primary efficiency correspondingly increased from 0.920 to 0.958, and thermodynamic efficiency decreased from 0.920 to 0.915. Thus, the stator cooling air did indeed contribute to turbine work output in passing through the rotor. The equivalent primary flow decreased slightly with the coolant flow admission.

The data in table II for turbine 3 summarize the test results previously discussed. This turbine had both stator- and rotor-blade trailing-edge coolant ejection slots. Turbine 3 was tested at an elevated turbine inlet temperature (378 K; 680° R) as compared to basically ambient temperature for turbines 1 and 2. Then, with stator-blade coolant flow and zero cooling air to the rotor (turbine 3a), the stator coolant fraction increased from 0.0468 to 0.0512 when compared with turbine 2b at comparable turbine operating conditions. This increase in coolant fraction is directly attributable to the primary-air inlet temperature. The equivalent primary flow rate also increased. Now, for all three turbines, the stator was unchoked and the rotor controlled the flow. Apparently, when slotting the trailing edges of the rotor blades (for turbine 3), the throat areas were increased, yielding the higher primary flow. The measured primary efficiency decreased from 0.958 to 0.933. Table II shows that less equivalent work and turning (outlet flow angle) were observed for turbine 3a. Thermodynamic efficiency decreased about

2 points (from 0.915 to 0.896). When the stator cooling efficiency was assumed to be the same as in reference 4, as discussed previously, the resultant efficiencies (footnote b in table II) for turbine 3a agreed very closely with results for turbine 2b.

Turbine 3b had cooling air supplied to both blade rows at turbine inlet pressure. The resultant stator coolant fraction was again 0.0512, as for turbine 3a. The rotor coolant fraction was 0.0638. The rotor fraction was higher, as stated previously, primarily because there were more slotted rotor blades than stator blades. To a lesser extent, this rotor coolant fraction could have been influenced by the added momentum of the coolant with rotor rotation (pumping) and because the rotor coolant discharged to a lower pressure than did the stator coolant. As noted in table II, the addition of rotor coolant did add to the turbine equivalent specific-work output (1.8 percent). Primary mass flow remained essentially the same. Outlet flow angle (turning) decreased 0.6°. The net effect on primary efficiency was a slight increase from the aforementioned calculated value of 0.954 to 0.955. Concomitantly, the thermodynamic efficiency decreased from 0.916 to 0.874 as a result of charging the turbine with the added ideal energy of the rotor coolant flow.

Turbine configurations 3b and 3c (table II) were identical. Data for turbine 3c were previously discussed herein when the overall performance of the turbine was determined over a range of speed and pressure ratio with coolant flow to both blade rows (see section Overall Turbine Performance). These data were obtained with a coolant supply temperature some 17 K (31° R) lower than that for turbine 3b and are included in the table for completeness. The lower coolant supply temperature for turbine 3c did result in coolant fraction increases since flow varies inversely as the square root of the temperature ratio. The net work output and efficiencies for turbine 3c are shown in table II to be slightly higher than those for turbine 3b.

Primary mass-flow rate. - Figure 19 presents the equivalent primary mass-flow rate as a function of pressure ratio at equivalent design speed for the turbine with solid (uncooled) blading (ref. 2), for the turbine with stator-blade coolant flow only (ref. 4), and for the subject turbine with both stator- and rotor-blade coolant flow. Cooling air, when applicable, was in all cases supplied at the turbine inlet pressure. At any comparable pressure ratio, the use of stator coolant flow resulted in a decrease in primary flow when compared with the uncooled turbine. The primary flow was significantly higher for the turbine with cooling air to both blade rows than for the other two turbines. This was noted when results from table II were discussed and is attributed to the fact that different rotor blading was used.

Turbine efficiency. - Figure 20 shows the variation of primary efficiency as a function of pressure ratio at the equivalent design speed and for the aforesaid three turbine configurations. Where stator coolant only was used, the coolant fraction was nominally 0.0468 (ref. 4). With cooling air to both blade rows, the coolant fractions were nominally 0.0524 for the stator and 0.0658 for the rotor (see fig. 11). Primary efficiency

was calculated as per equation (3) wherein the equivalent ideal work output was based on the primary flow.

The rise in primary efficiency with stator cooling (fig. 20) resulted from the increased torque output as the cooling air developed work in passing through the rotor (ref. 4). The further addition of rotor cooling air resulted in no significant change in efficiency except at pressure ratios less than about 1.7. In this regime an actual performance penalty resulted with rotor coolant addition.

Blade-row reaction characteristics. - Blade-row reaction is presented in figure 21 in terms of static-pressure variations through the turbine as measured at both the hub and the tip for the three different turbine configurations. Data were normalized by dividing individual pressures by their corresponding turbine inlet total pressures. The data shown in figure 21 correspond to a turbine pressure ratio of 1.755 and were obtained for each turbine from plots such as that shown in figure 16. It is significant that at this pressure ratio all three turbines were unchoked (fig. 19).

Stator reaction: As shown in table II the primary flow for the turbine with only stator cooling decreased as compared with the uncooled turbine. This lower primary flow, in turn, decreased the reaction across the stator (fig. 21). With cooling air to both blade rows, the primary flow increased but had little effect on the stator-hub reaction. At the stator blade tip, the stator reaction increased but was less than that observed for the uncooled turbine.

Rotor reaction: Adding only stator coolant flow increased the reaction across the rotor (fig. 21) at both the hub and the tip. With coolant flow through both blade rows, the reaction across the rotor at the hub increased. At the tip, however, no significant change in reaction was noted.

Even though the total-pressure ratio was the same for all three turbines, exit static pressures varied. This difference results from the fact that the total flow (primary plus coolant) was used to calculate the exit total pressure (eq. (2)).

SUMMARY OF RESULTS

A single-stage, 0.762-meter- (30-in.-) tip-diameter turbine was tested to determine the effect on its aerodynamic performance of coolant ejection from slots in the trailing edges of both stator and rotor blades. The experimental results include the efficiency and mass-flow characteristics over a range of speed and overall pressure ratio. Coolant flow to the rotor was also independently varied at a given turbine operating point. The results are compared with those previously obtained for the same basic turbine but having (1) uncooled (solid) blading and (2) only stator-blade cooling-air ejection. All tests were conducted at, or near, a primary- to coolant-air temperature ratio of unity. Pertinent findings are summarized as follows:

1. When operated over a wide range of speed and pressure ratio with coolant air supplied to both blade rows at a pressure equal to turbine inlet pressure, the turbine exhibited high primary efficiency (defined herein). Efficiencies greater than 0.96 were noted at 110-percent speed and low pressure ratios.

2. At equivalent design speed, a pressure ratio of 1.755, and with coolant air supplied to both blade rows at a pressure equal to the turbine inlet pressure, the primary efficiency was 0.958, the peak value obtained at the equivalent design speed. The thermodynamic efficiency was 0.878. Attendant stator and rotor coolant fractions were 0.0524 and 0.0658, respectively. The primary efficiency was identical to that obtained when the turbine with only stator cooling was tested. The corresponding thermodynamic efficiency for this turbine configuration was 0.915. The primary efficiency for the uncooled turbine configuration was 0.923.

3. At equivalent design speed, a pressure ratio of 1.755, and with coolant air supplied to only the stator blades at turbine inlet pressure (a coolant fraction of 0.0512), adding rotor coolant imposed a severe penalty on turbine efficiency. Thermodynamic efficiency decreased linearly with rotor coolant at a rate of about 0.7 percent per percent rotor coolant fraction. It required a rotor coolant fraction above 0.063 before a net improvement in torque output (and primary efficiency) was realized.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, January 8, 1975,
505-04.

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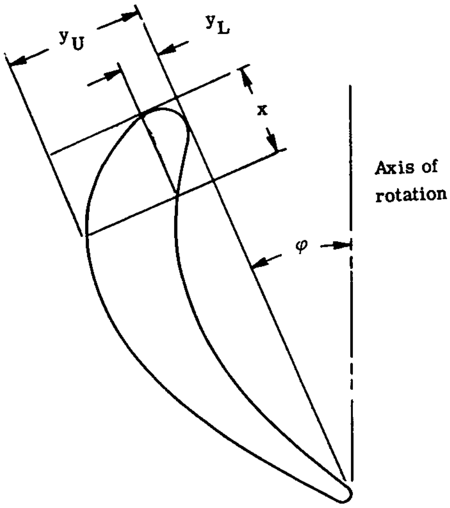
TABLE I. - FIRST-STAGE-TURBINE BLADE COORDINATES

(a) Stator[a]

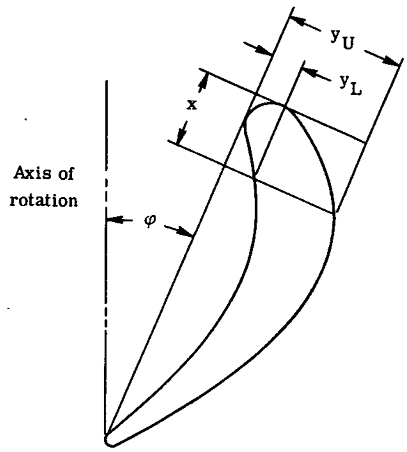


x		Hub				Mean				Tip			
		Orientation angle, φ, deg											
		42.42				41.03				39.67			
		y_L		y_U		y_L		y_U		y_L		y_U	
cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.
0	0	0.381	0.150	0.381	0.150	0.381	0.150	0.381	0.150	0.381	0.150	0.381	0.150
.254	.100	-----	-----	.953	.375	-----	-----	1.001	.394	-----	-----	1.085	.427
.508	.200	-----	-----	1.234	.486	-----	-----	1.306	.514	-----	-----	1.397	.550
.762	.300	.152	.060	1.417	.558	.155	.061	1.473	.588	.160	.063	1.577	.621
1.016	.400	.267	.105	1.532	.603	.269	.106	1.615	.636	.282	.111	1.689	.665
1.270	.500	.363	.143	1.600	.630	.368	.145	1.689	.665	.376	.148	1.748	.688
1.524	.600	.442	.174	1.633	.643	.442	.174	1.717	.676	.455	.179	1.768	.696
1.778	.700	.500	.197	1.633	.643	.498	.196	1.716	.675	.516	.203	1.753	.690
2.032	.800	.544	.214	1.613	.635	.533	.210	1.684	.663	.551	.217	1.715	.675
2.286	.900	.574	.226	1.570	.618	.556	.219	1.636	.644	.577	.227	1.654	.651
2.540	1.000	.584	.230	1.511	.595	.566	.223	1.572	.619	.587	.231	1.588	.625
2.794	1.100	.579	.228	1.448	.570	.561	.221	1.499	.590	.582	.229	1.516	.597
3.048	1.200	.566	.223	1.374	.541	.546	.215	1.422	.560	.584	.223	1.435	.565
3.302	1.300	.538	.212	1.290	.508	.521	.205	1.339	.527	.544	.214	1.359	.535
3.556	1.400	.498	.196	1.201	.473	.485	.191	1.250	.492	.508	.200	1.270	.500
3.810	1.500	.445	.175	1.100	.433	.445	.175	1.148	.452	.465	.183	1.173	.462
4.064	1.600	.389	.153	.787	.391	.394	.155	1.041	.410	.414	.163	1.072	.422
4.318	1.700	.325	.128	.876	.345	.338	.133	.927	.365	.356	.140	.965	.380
4.572	1.800	.262	.103	.749	.295	.282	.111	.810	.319	.297	.117	.851	.335
4.826	1.900	.191	.075	.615	.242	.218	.086	.678	.267	.241	.095	.729	.287
5.080	2.000	.117	.046	.465	.183	.152	.060	.544	.214	.178	.070	.602	.237
5.334	2.100	.041	.016	.307	.121	.084	.033	.399	.157	.114	.045	.470	.185
5.580	2.197	.089	.035	-----	-----	-----	-----	-----	-----	-----	-----	-----	-----
5.588	2.200	-----	-----	-----	-----	.013	.005	.244	.096	.053	.021	.330	.130
5.748	2.263	-----	-----	-----	-----	.089	.035	.089	.035	-----	-----	-----	-----
5.911	2.327	-----	-----	-----	-----	-----	-----	-----	-----	.089	.035	.089	.035

[a]From ref. 5, revised.

TABLE I. - Concluded. FIRST-STAGE-TURBINE BLADE COORDINATES

(b) Rotor[a]



x		Hub				Mean				Tip			
		Orientation angle, φ, deg											
		11.31				22.87				34.67			
		y_L		y_U		y_L		y_U		y_L		y_U	
cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.	cm	in.
0	0	0.381	0.150	0.381	0.150	0.381	0.150	0.381	0.150	0.381	0.150	0.381	0.150
.254	.100	-----	-----	.998	.393	-----	-----	.909	.358	-----	-----	.792	.312
.508	.200	-----	-----	1.361	.536	-----	-----	1.254	.490	-----	-----	1.008	.397
.762	.300	.191	.075	1.631	.642	.198	.078	1.506	.593	.168	.066	1.189	.468
1.016	.400	.378	.149	1.842	.725	.389	.153	1.697	.668	.310	.122	1.328	.523
1.270	.500	.556	.219	2.002	.788	.551	.217	1.834	.722	.427	.168	1.435	.565
1.524	.600	.719	.283	2.131	.839	.678	.267	1.918	.755	.518	.204	1.506	.593
1.778	.700	.859	.338	2.223	.875	.780	.307	1.966	.744	.589	.232	1.549	.610
2.032	.800	.973	.383	2.286	.900	.861	.339	1.984	.781	.635	.250	1.560	.614
2.286	.900	1.057	.416	2.319	.913	.914	.360	1.969	.775	.665	.262	1.547	.609
2.540	1.000	1.115	.439	2.319	.913	.947	.373	1.928	.759	.671	.264	1.509	.594
2.794	1.100	1.151	.453	2.291	.902	.958	.377	1.864	.734	.663	.261	1.455	.573
3.048	1.200	1.163	.458	2.255	.878	.947	.373	1.783	.702	.640	.252	1.389	.547
3.302	1.300	1.153	.454	2.146	.845	.919	.362	1.687	.664	.602	.237	1.318	.519
3.556	1.400	1.123	.442	2.035	.801	.869	.342	1.575	.620	.559	.220	1.237	.487
3.810	1.500	1.072	.422	1.900	.748	.842	.315	1.455	.573	.508	.200	1.151	.453
4.064	1.600	.998	.393	1.750	.689	.719	.283	1.328	.523	.450	.177	1.059	.416
4.318	1.700	.904	.356	1.585	.624	.620	.244	1.184	.466	.389	.153	.958	.377
4.572	1.800	.792	.312	1.410	.555	.516	.203	1.034	.407	.325	.128	.846	.333
4.826	1.900	.660	.260	1.220	.481	.404	.159	.879	.346	.262	.103	.732	.288
5.080	2.000	.513	.202	1.016	.400	.287	.113	.704	.277	.196	.077	.605	.238
5.334	2.100	.356	.140	.792	.312	.170	.067	.518	.204	.127	.050	.470	.185
5.588	2.200	.183	.072	.549	.216	.056	.022	.318	.125	.056	.022	.325	.128
5.817	2.290	-----	-----	-----	-----	.089	.035	.089	.035	-----	-----	-----	-----
5.842	2.300	-----	-----	.279	.110	-----	-----	-----	-----	-----	-----	-----	-----
5.895	2.321	-----	-----	-----	-----	-----	-----	-----	-----	.089	.035	.089	.035
5.979	2.354	.089	.035	.089	.035	-----	-----	-----	-----	-----	-----	-----	-----
		Stacking axis coordinates											
		x = 3.048(1.200)		y = 1.019(0.401)		x = 2.819(1.110)		y = 0.955(0.376)		x = 2.743(1.080)		y = 0.856(0.337)	

[a]From ref. 5, revised.

TABLE II. - PERFORMANCE COMPARISON OF UNCOOLED AND COOLED TURBINES

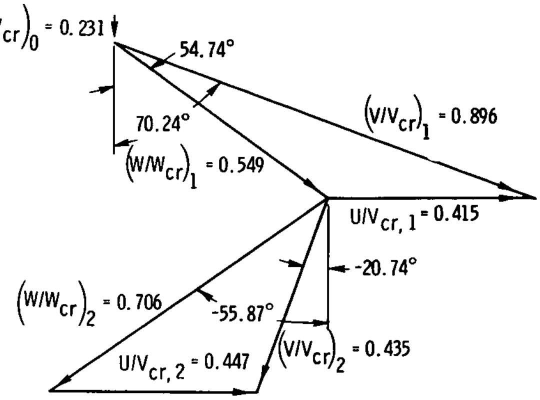
Turbine	Pressure ratio	Equivalent work		Equivalent primary flow		Outlet flow angle, deg	Coolant fraction[a]		Efficiency		Turbine inlet temperature		Nominal coolant inlet temperature	
		J/g	Btu/lbm	kg/sec	lbm/sec		Stator	Rotor	Primary	Thermo-dynamic	K	°R	K	°R
1 - Design (solid blades, ref. 2)	1.751	39.57	17.00	18.43	40.64	-15.2	------	------	0.923	0.923	303	545	---	---
2a - Stator slotted, no cooling air (ref. 4)	1.755	39.57	17.00	18.38	40.54	-15.7	0	------	0.920	0.920	303	545	---	---
2b - Stator slotted, cooling air (ref. 4)	1.755	41.30	17.74	18.13	39.96	-18.3	0.0468	------	0.958	0.915	303	545	303	545
3a - Stator and rotor slotted, cooling air to stator	1.755	40.36	17.34	18.61	41.02	-14.5	0.0512	0	0.933 [b].954	0.896 [b].916	378	680	304	548
3b - Stator and rotor slotted, cooling air to both	1.755	41.13	17.67	18.67	41.16	-13.9	0.0512	0.0638	0.955	0.874	378	680	304	548
3c - Stator and rotor slotted, cooling air to both (at lower temperature)	1.755	41.23	17.71	18.60	41.01	-14.6	0.0524	0.0656	0.958	0.878	378	680	287	517

[a]Conditions where coolant supply pressure (or pressures) equals turbine inlet pressure.
[b]Stator coolant efficiency assumed same as in ref. 4.

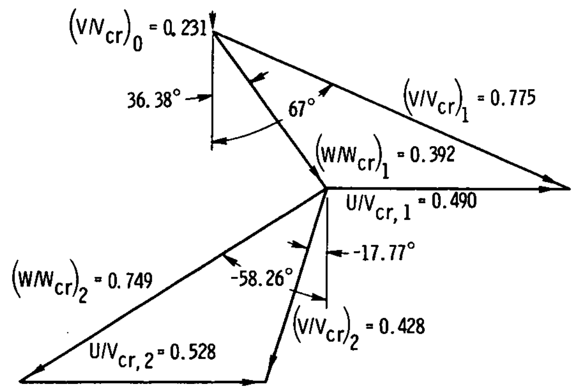
FIRST-STAGE TURBINE VELOCITY DIAGRAM VECTOR VALUES

[All velocities based on turbine inlet conditions of U. S. standard sea-level air.]

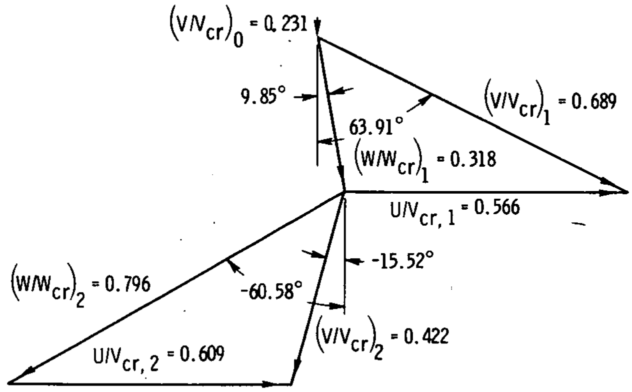
Vector	Hub	Mean	Tip
	Velocity, m/sec (ft/sec)		
U_1, U_2	129.0 (423.1)	152.4 (500)	175.8 (576.9)
$V_{u,1}$	262.1 (859.8)	221.8 (727.6)	192.2 (630.5)
$V_{u,2}$	44.5 (146.1)	37.7 (123.6)	32.7 (107.2)
$V_{x,1}$	94.1 (308.8)	94.1 (308.8)	94.1 (308.8)
$V_{x,2}$	117.6 (385.8)	117.6 (385.8)	117.6 (385.8)
V_1	278.5 (913.6)	240.9 (790.4)	214.0 (702.1)
V_2	125.7 (412.5)	123.5 (405.1)	122.0 (400.4)
W_1	163.0 (534.9)	116.9 (383.6)	95.5 (313.4)
W_2	209.6 (687.6)	223.5 (733.3)	239.4 (785.4)



(a) Hub section; radius ratio, r/r_t, 0.733.

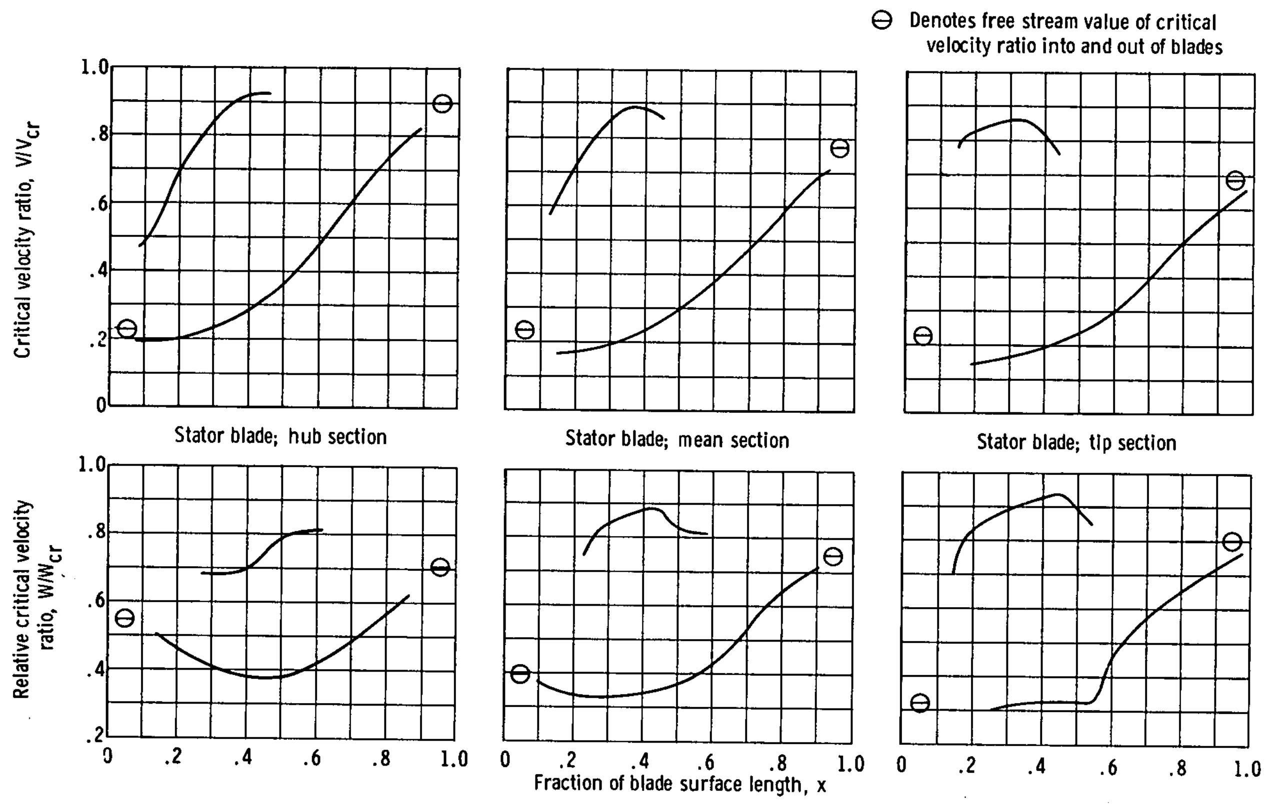


(b) Mean section; radius ratio, r/r_t, 0.8666.



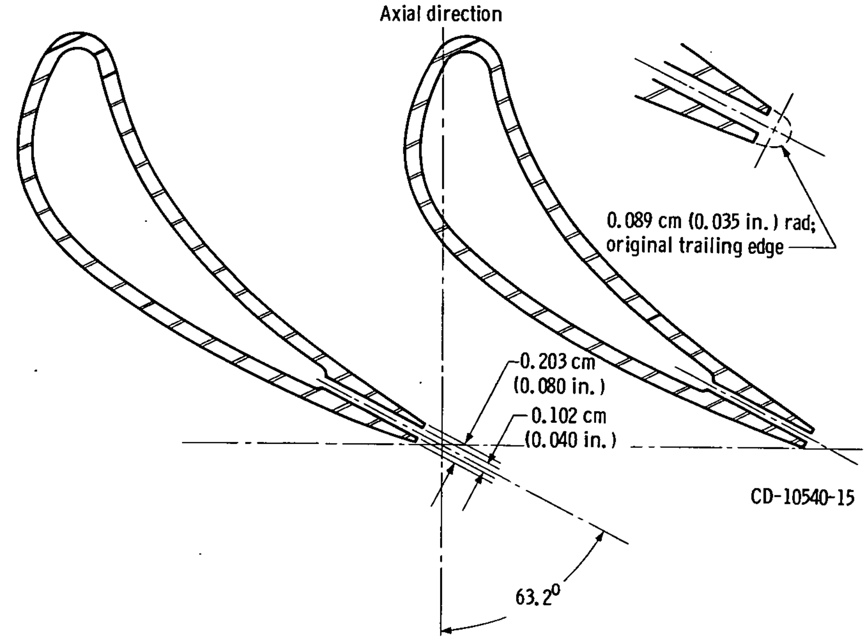
(c) Tip section; radius ratio, r/r_t, 1.000.

Figure 1. - First-stage-turbine design velocity diagram. (From ref. 5.)

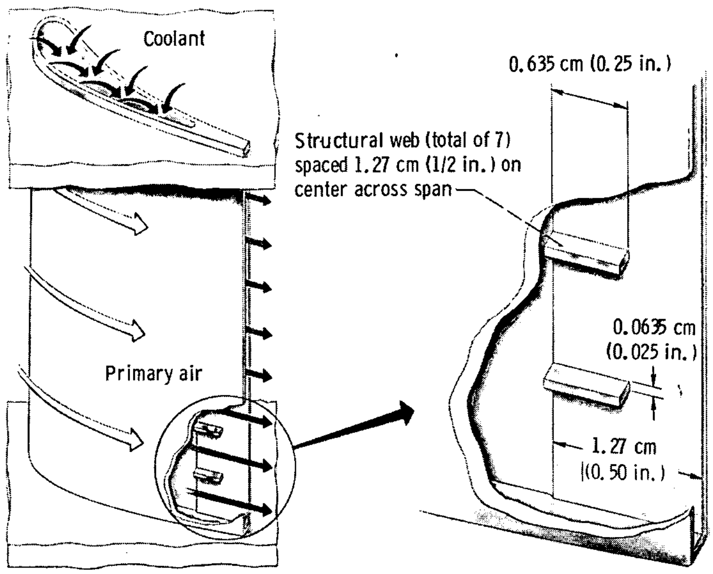


(b) First stage.

Figure 2. - First-stage-turbine design blade-surface velocity distributions. (From ref. 5.)



(a) Cross-sectional view.



(b) Cutaway view.

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Figure 3. - Modified design stator blades.

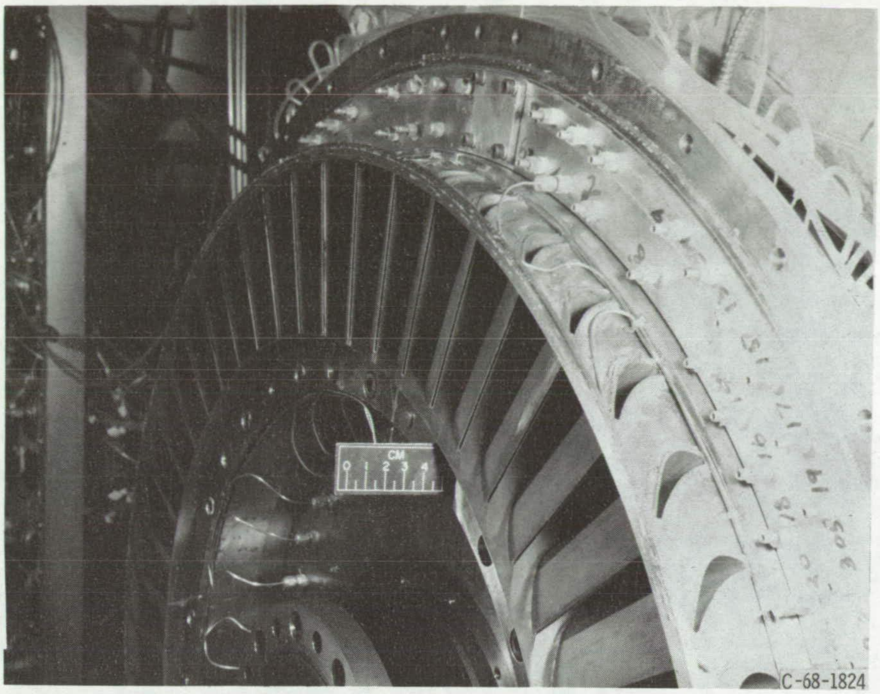

Figure 4. - Closeup of slotted stator-blade assembly.

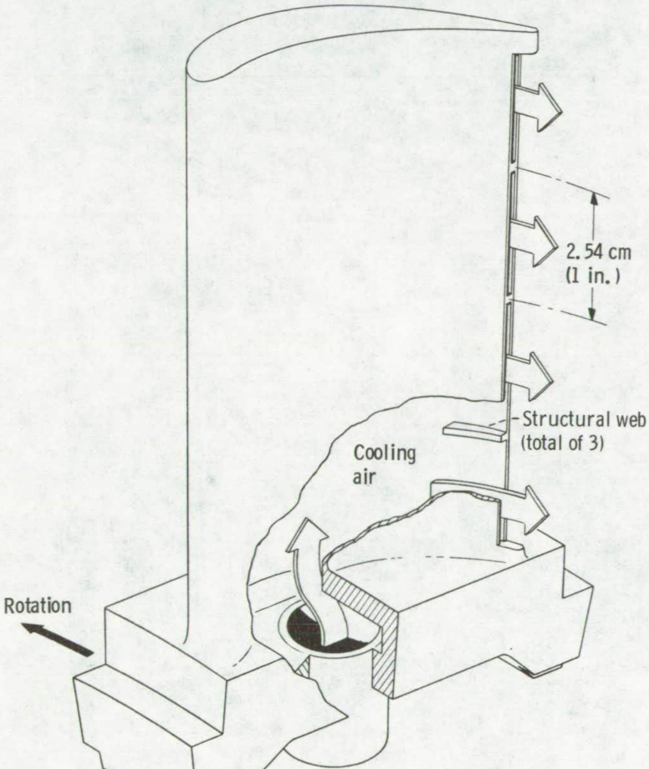

Figure 5. - Rotor blade.

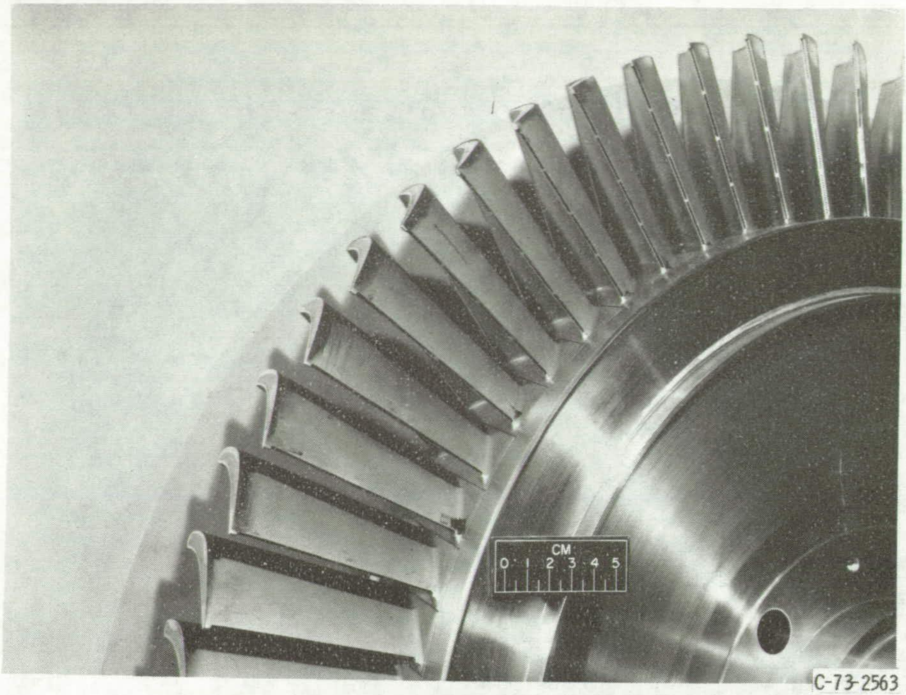

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Figure 6. - Closeup of slotted rotor-blade assembly.

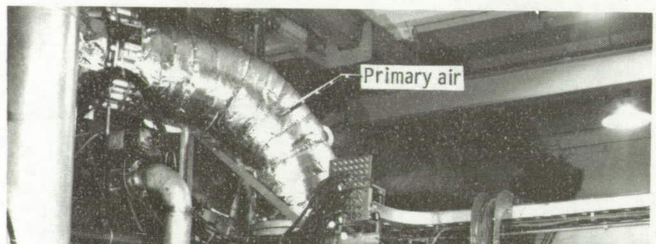

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Figure 7. - Test facility.

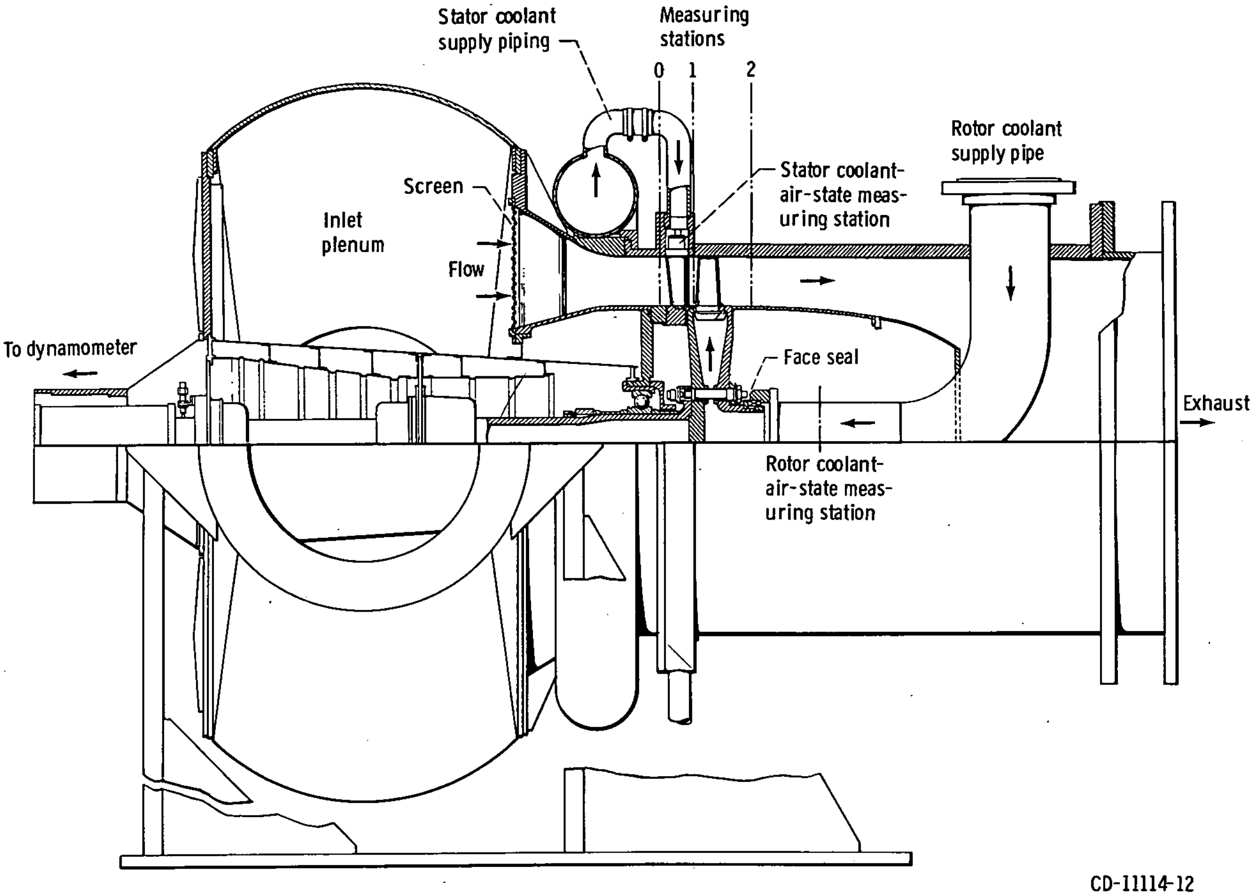


Figure 8. - Cross-sectional view of turbine test section.

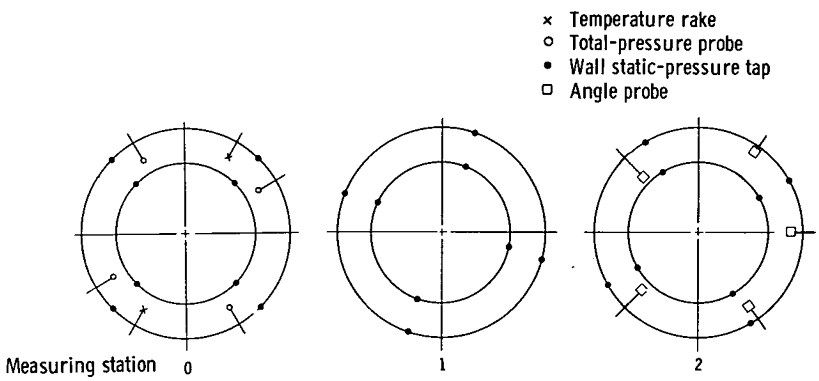


Figure 9. - Schematic diagram of turbine instrumentation, viewed upstream.

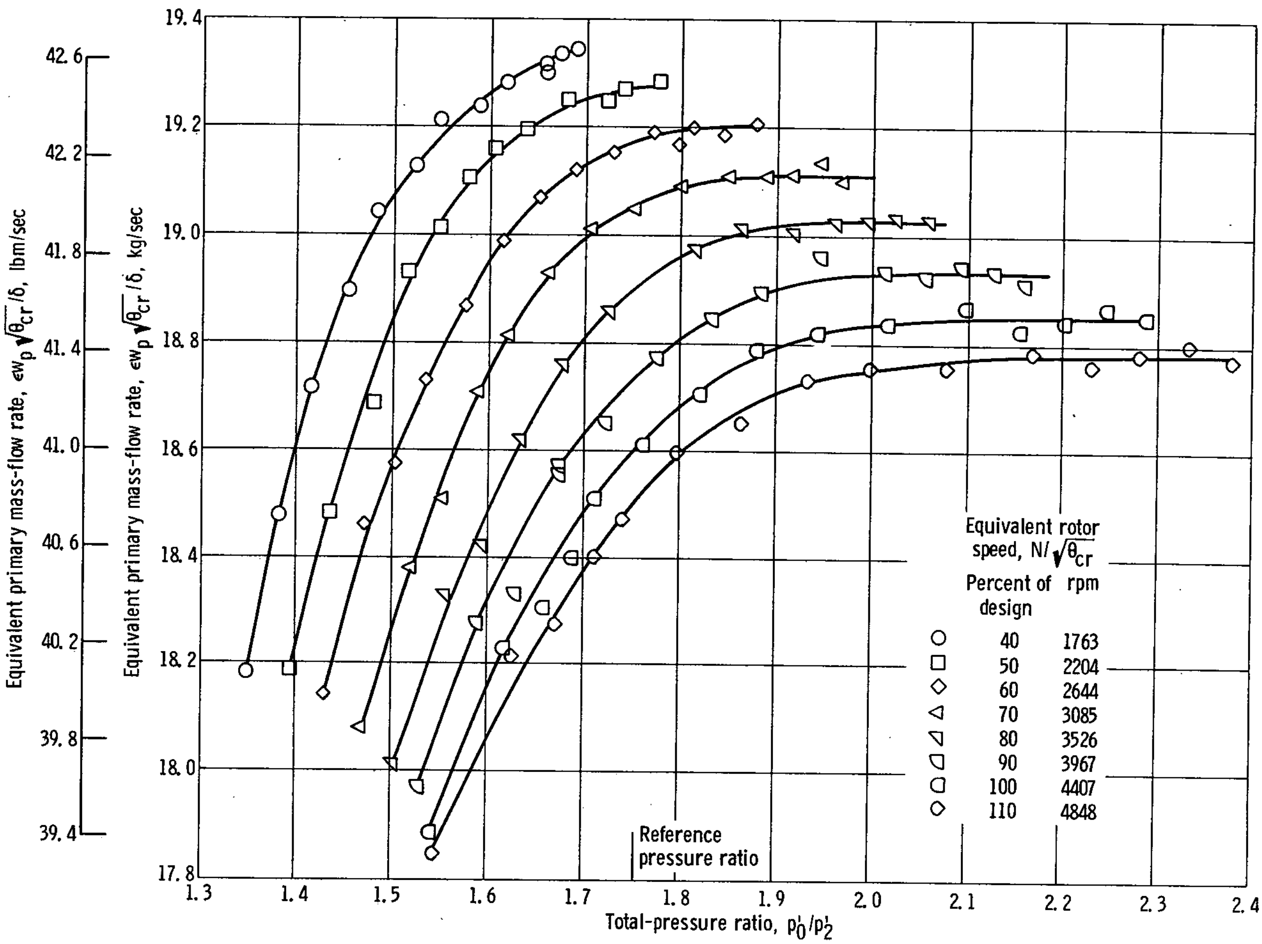


Figure 10. - Variation of equivalent primary mass-flow rate with total-pressure ratio and equivalent rotor speed. Cooling air supplied at turbine inlet pressure.

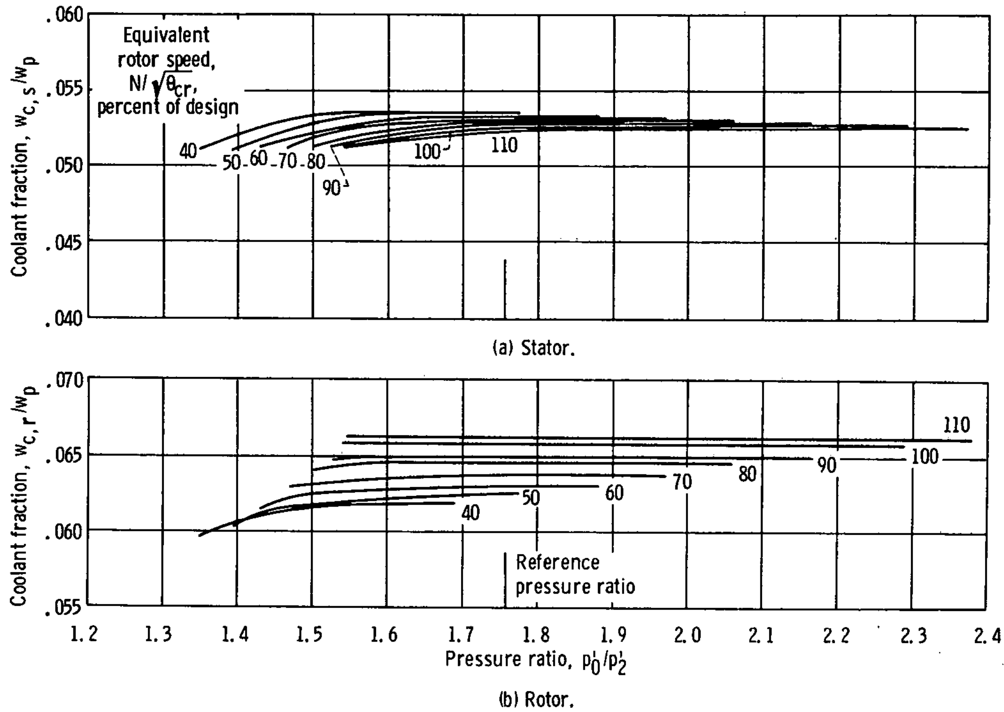


Figure 11. - Variation of faired coolant fractions with equivalent speed and pressure ratio.

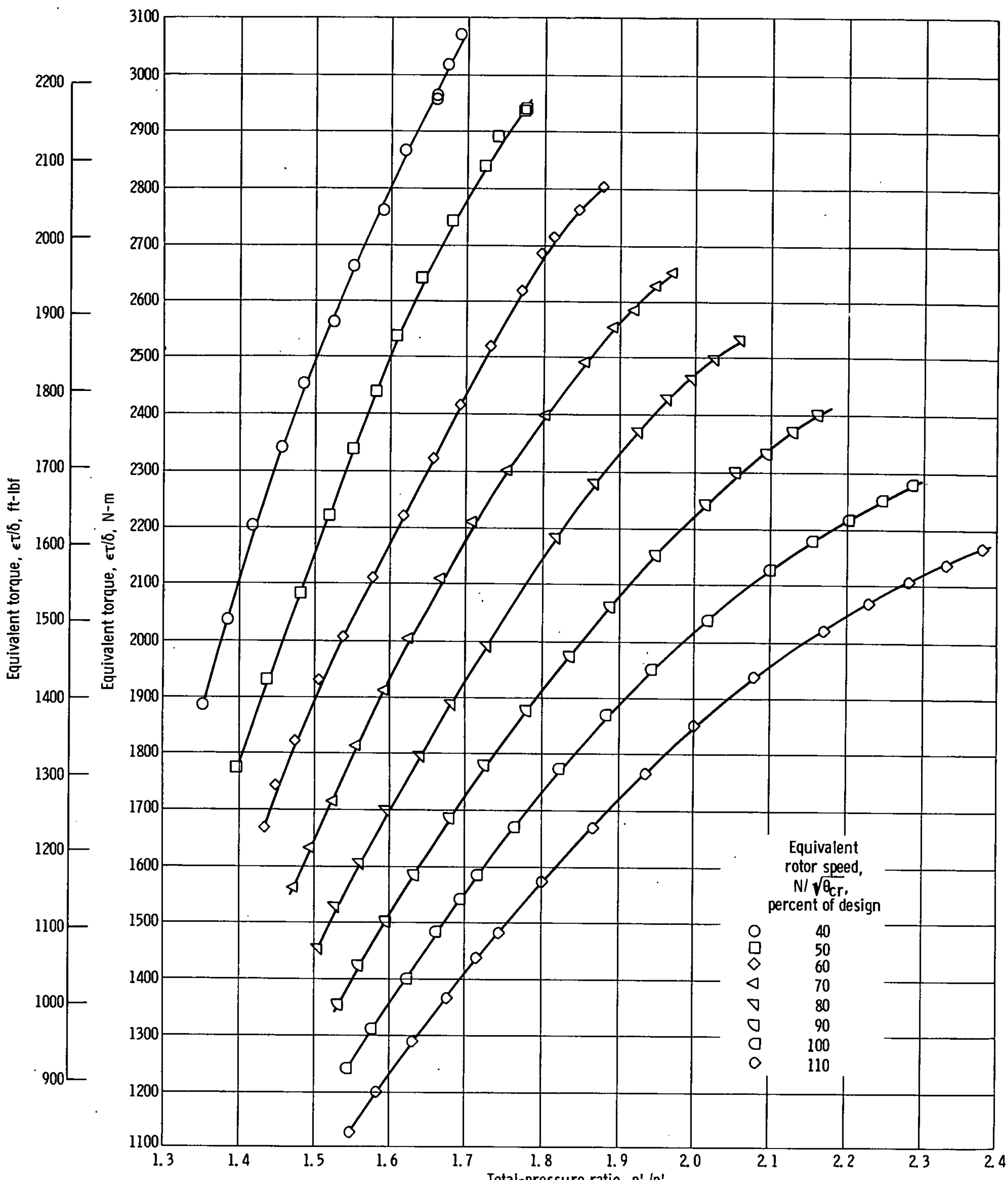


Figure 12. - Variation of equivalent torque with total-pressure ratio and equivalent rotor speed. Cooling air supplied at turbine inlet pressure.

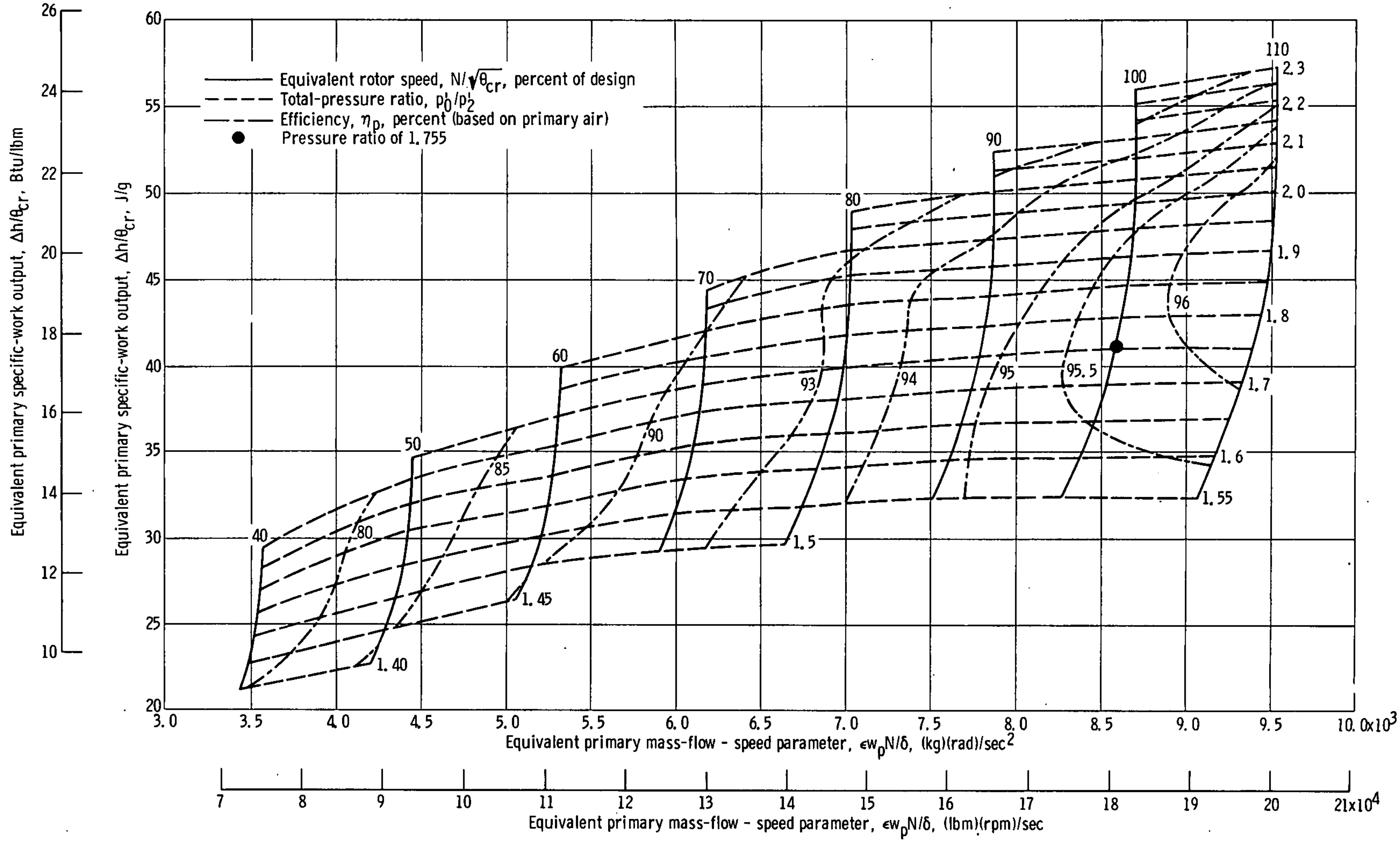


Figure 13. - Overall performance map of cooled turbine. Cooling air supplied at turbine inlet pressure.

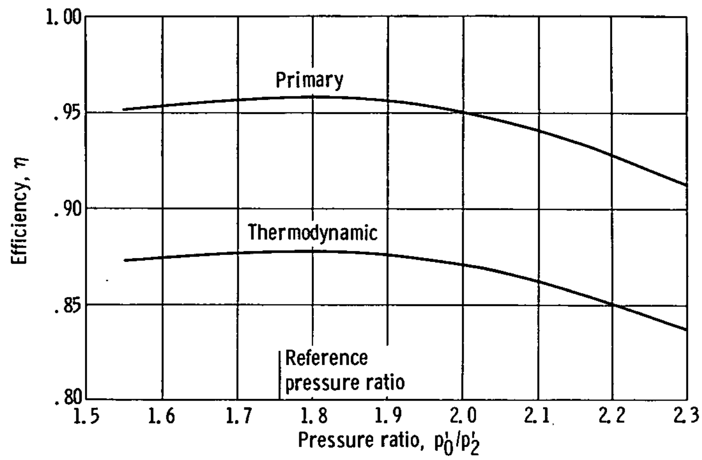


Figure 14. - Variation of primary and thermodynamic efficiencies with pressure ratio for equivalent design speed. Cooling air supplied at turbine inlet pressure.

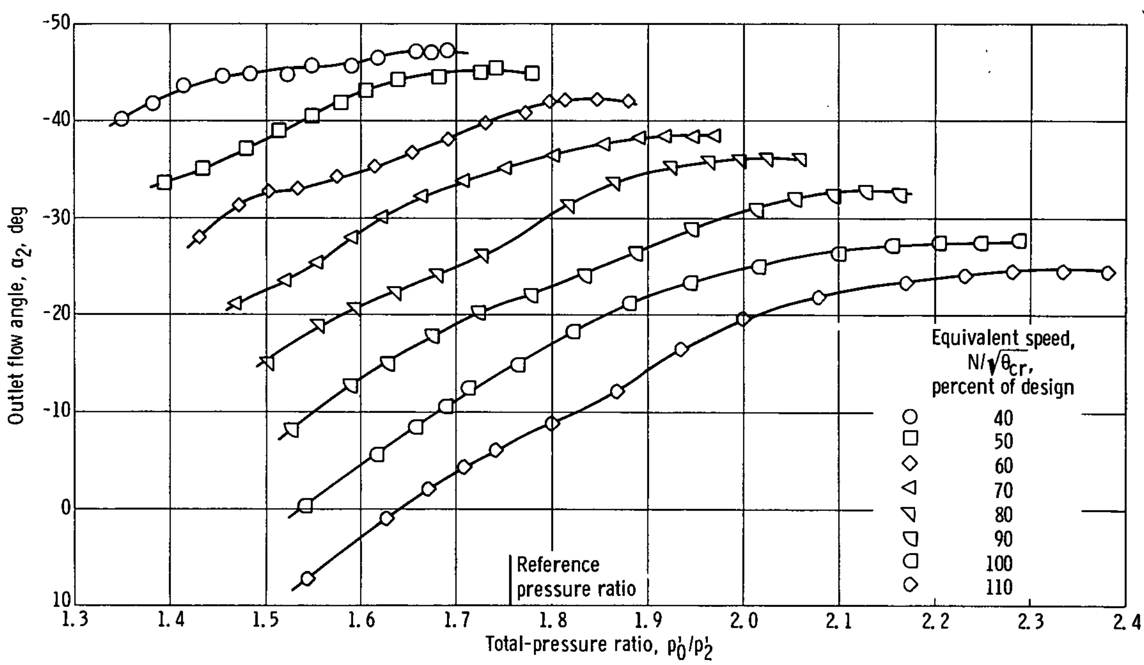
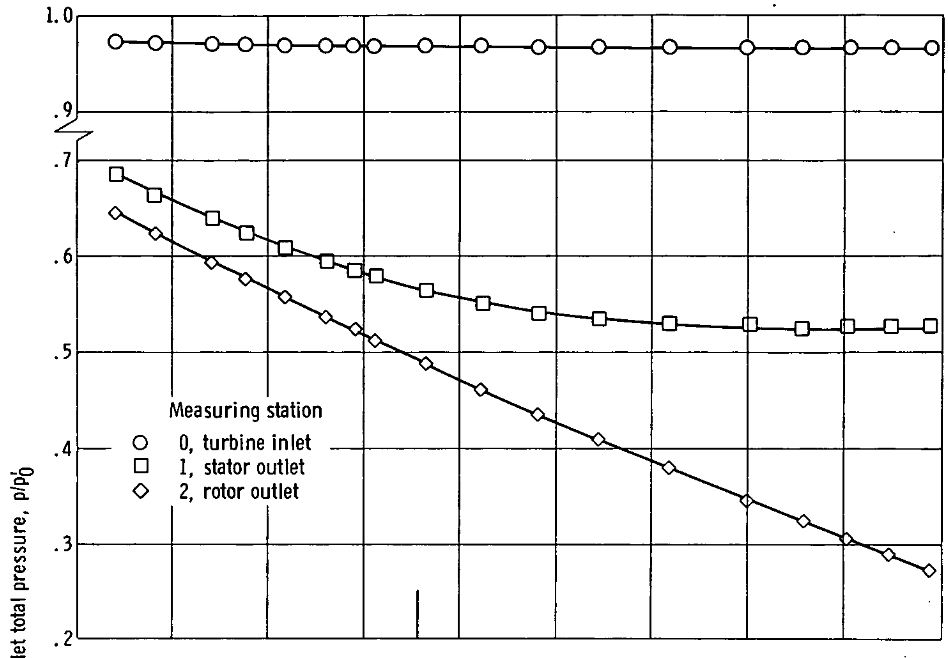
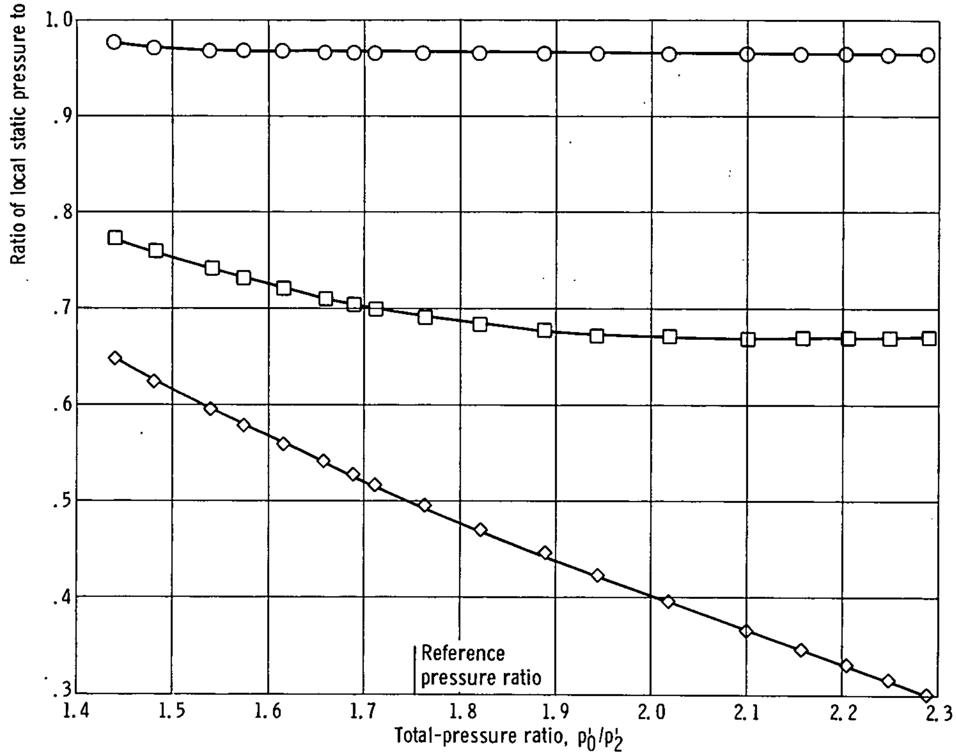


Figure 15. - Variation of outlet flow angle with total-pressure ratio and equivalent rotor speed. Cooling air supplied at turbine inlet pressure.



(a) Hub.



(b) Tip.

Figure 16. - Variation of static pressure through turbine with total-pressure ratio at equivalent design speed. Cooling air supplied at turbine inlet pressure.

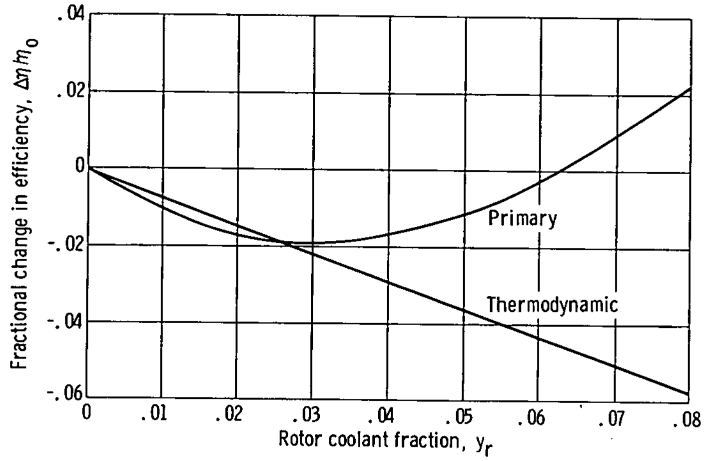


Figure 17. - Effect of rotor coolant on primary and thermodynamic efficiencies. Data at equivalent design speed and a pressure ratio of 1.755; stator coolant fraction, y_s, 0.0512.

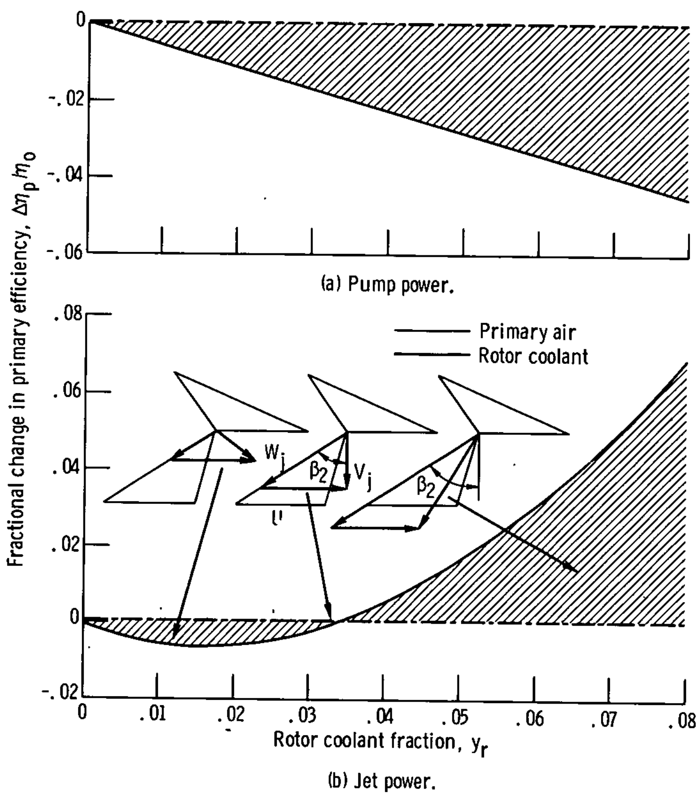


Figure 18. - Effect of pump and jet power on primary efficiency. Data at equivalent design speed and a pressure ratio of 1.755; stator coolant fraction, y_s, 0.0512.

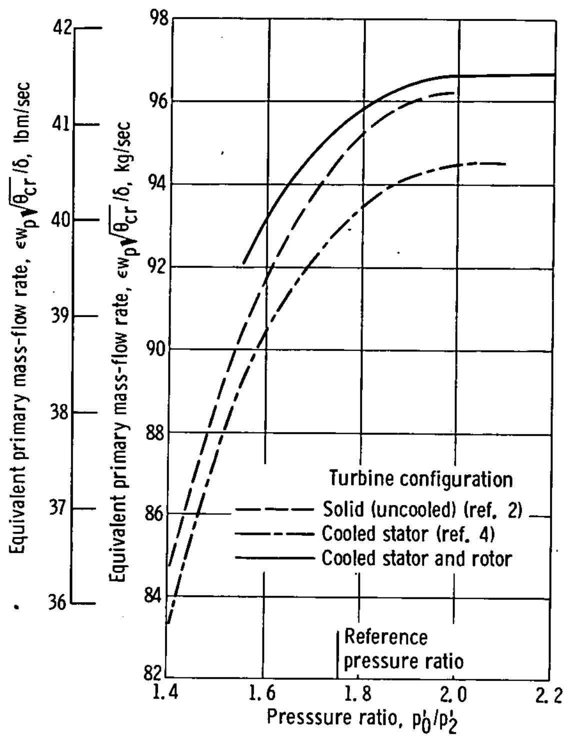


Figure 19. - Variation of equivalent primary mass-flow rate with pressure ratio at equivalent design speed for three turbine configurations. Cooling air supplied at turbine inlet pressure.

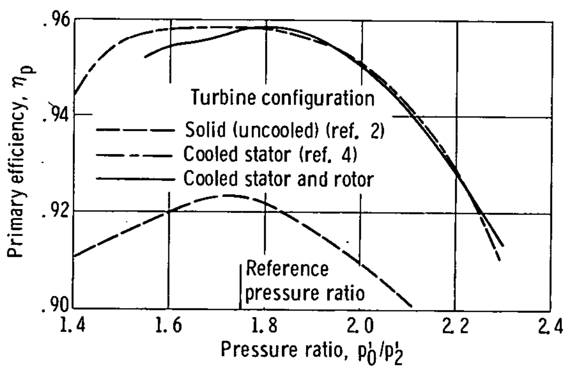


Figure 20. - Variation of primary efficiency with pressure ratio at equivalent design speed for three turbine configurations. Cooling air supplied at turbine inlet pressure.

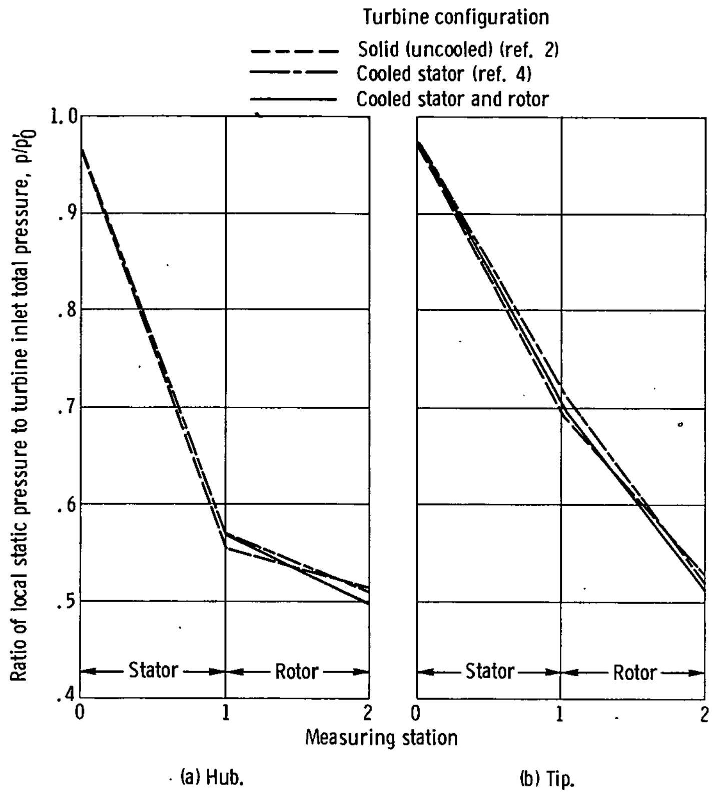


Figure 21. - Comparison of static-pressure variation through turbines at equivalent design speed and a pressure ratio of 1.755. Cooling air supplied at turbine inlet pressure.

NASA-Langley, 1975 E-8158

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—NATIONAL AERONAUTICS AND SPACE ACT OF 1958

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